ROTATING MACHINERY ENERGY LOSS DUE TO MISALIGNMENT

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ABSTRACT:
Tests were conducted to measure the energy lost due to misalignment and unbalance of rotating machinery. Several articles have appeared in maintenance publications that warn of misalignment energy losses of up to 15 percent. Energy losses of this a magnitude would dictate major machinery maintenance upgrading, precision alignment, and use of vibration and infrared diagnostics. Controlled tests were conducted to measure energy losses due to misalignment and unbalance. The greatest loss measured was 1.2 percent at 25 percent power for gross misalignment; most losses were far less. Losses due to unbalance were under half of that. The tests were conducted with a 30-hp, 3-phase motor driving a 20-kW generator connected to a resistive load bank. Power into the motor and from the generator to the load was measured with precision power measurement instrumentation. Four popular couplings were tested. This paper describes the tests, presents data, and comments on the ramifications.

KEY WORDS: Energy loss; misalignment; rotating machinery; unbalance.

INTRODUCTION:
Machinery vibration problems cause additional loads on bearings, seals, couplings, and foundations and thus must be suspected of causing excess power consumption. Some of the literature is reporting that poorly balanced and misaligned machines are consuming up to 15 percent extra energy. Moore, Pardue, and Piety [1] report "Eliminating high energy vibration sources such as misalignment and imbalance can reduce machine power consumption 10 to 15 percent." An analysis by Xu, Zatezalo, and Marangoni from the University of Pittsburgh [2] yielded a heavy power loss for angular misalignment up to five degrees; in the case of one degree, it was 114 kWh/hp-yr. Reference [3] cites energy savings of 11 percent, attributed somewhat to a data collector manufacturer's literature. Their presentation discussed measurements they have taken showing energy reductions of up to 60 percent. They claim to routinely find savings of 10 percent, with a minimum of 5 percent. Reference [4] discusses measurements that show a power loss of 2.3 percent for a loaded machine and 9.1 percent for an unloaded machine.

Additionally, experimental work that shows that the couplings consume energy was performed by Bortnem, Pray, and Grover of the Miller Brewing Company [5]. They connected a 10-hp motor to an 8-kW generator loaded to 7.6 kW and measured coupling temperatures as a function of misalignment. They found a linear coupling temperature rise proportional to parallel misalignment with a maximum of 41 ^{o}F for a 0.030-inch misalignment. When the alignment was as perfect as it could be measured with a laser ambient.

Recently, reference [6] presented results of a University of Tennessee study that essentially finds no energy loss due to misalignment. The reference indicates a test procedure similar to ours except with one different coupling and a different machine foundation. They confined the amount of misalignment to the limits that the manufacturers recommend, whereas we considered much greater misalignments, as much as four times the manufacturers recommendations.

Energy losses of even 5 percent would cause major machinery maintenance changes, precision alignment and balancing, and wider use of vibration and infrared diagnostics. Due to these possibilities, the Navy sponsored an experimental study to scientifically determine if these energy losses could be confirmed. For those unfamiliar with conditions that give rise to gross misalignment, it can arise during installation whenever considerable care is not expended to assure its absence. In other words, it is a great deal of trouble to precision align new machines.

The Experimental Program:
An experimental program was established to conduct accurate measurements of the energy lost due to misalignment and unbalance of rotating machinery. The tests were accomplished with a 30-hp, 3-phase motor driving a 20-kW, 400-cycle AC generator connected to a resistive load bank. Various couplings were used. Power to the motor and from the generator to the load bank was accurately measured. Thus an efficiency could be calculated. Changes in the efficiencies were used to indicate losses from unbalance and misalignment.

Power into the motor and from the generator to the load was measured with precision power measurement instrumentation, consisting of precision digital wattmeters, and precision current transformers. The Appendix lists the test equipment. Measuring the efficiencies, as accurately as this set up permitted, made apparent the increasing thermal copper resistance losses due to temperature rise in the generator. This ultimately required that data only be taken after the temperature rise in the generator had stabilized.

Temperatures were measured with thermocouples and, at times, with an infrared camera. These were needed to warn of possible damage to the machines and to evaluate the misalignment generated by warming of the machines. Additionally, a thermocouple was bonded to the stator of the generator to monitor internal heat build up. This was the main sensor used to judge when a stabilized running condition had been reached. Data collection was taken hourly until this temperature stopped rising.

Four popular couplings were selected for testing. These are the grid coupling, the rubber boot coupling, a gear coupling, and a disk coupling. These were selected after considerable discussions with experts in the field. Manufacturers and sizes are given in Table I.

Test Machine Foundation:
Considerable thought was given to the machine's foundation. We thought that mechanisms for energy loss existed and were associated with deteriorated foundations that would allow motions with rubbing and friction losses, or from foundations connected to the earth or structure in such a way that allows energy to radiate into the surrounding ground or structure. A study of foundation energy loss mechanisms was beyond our scope. To eliminate these energy loss possibilities, an extremely rigid foundation was designed and fabricated. To attain extreme torsional rigidity, the typical wide flange frame was augmented by welding a heavy plate to both the top and bottom of the frame. Figure 1 is photograph of the completed foundation. The motor foundation to shaft height was 3.5 inches shorter than that of the generator; the 3.5 inch height difference was corrected by welding a frame of 2-inch box beams to the base plate and then welding a 1-inch plate to the box beams. To eliminate soft foot problems and facilitate accurate alignment and misalignment, we machined the feet of the motor as well as the heavy plate of the rigid foundation to which it was bolted.

The rigidity of this foundation raised concerns. When measuring vibration velocity spectra to note the symptoms of the misalignment and unbalance introduced, the levels were surprisingly small, even with almost half a pound of weight attached. The levels were inconsistent with levels measured in the field on commercial foundations. It was felt that a commercial foundation should also be tested since, being more flexible, it would allow more motion at the coupling and possibly consume more energy. A commercial foundation was fabricated as shown in Figure 2. This was used for the remaining tests since it permitted vibration levels consistent with field practice. However, it had little effect on the energy consumed due to unbalance or misalignment.

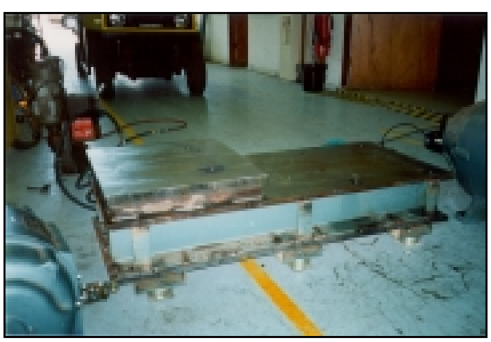

Figure 1. Rigid foundation.

Figure 2. Commercial foundation.

Data Analysis Procedures:
We made several beginnings with these measurements. As discussed, the concept was to use precision current transformers feeding precision wattmeters with a motor driving a

generator into a resistive load bank. Efficiencies were calculated as the output power divided by the input power or:

$$\eta = \frac{P_{output}}{P_{input}} \qquad (1)$$

Interestingly, the efficiency was affected by generator temperature; it would fluctuate considerably as the generator warmed up, decreasing to a steady state as the generator temperature stabilized. This took from 2 to 3 hours.

Power measurements were made with the generator fully loaded, 25 percent loaded, and unloaded. A lightly loaded machine might waste a greater fraction of its consumed power on parasitic effects such as coupling and bearing losses. Power measurements were made by manually recording input (to the motor) and output power (from the generator to the resistive load bank) from the precision wattmeters after the generator stator temperatures had stabilized. Output power was divided by input power to form efficiencies, except in the no-load tests. Efficiencies so calculated from measurements were compared for the various misalignment and unbalanced conditions. Efficiencies were all under 80 percent. Changes in efficiencies due to misalignment and unbalance were calculated as:

$$\Delta\eta = \frac{\eta_{undisturbed} - \eta_{disturbed}}{\eta_{undisturbed}} \qquad (2)$$

The only way increased energy consumption of consequence can occur is if excess energy consumption continues over a relatively long period of time. An unlubricated or sand loaded gear or grid coupling may consume more energy than a lubricated one for a short period of time, however, it would soon fail and cease to consume energy. Thus we made no tests of an unlubricated or sand impregnated grid or gear coupling.

Misalignment Procedures:
Shaft misalignment is the deviation of the driving and driven relative shaft centerline position from a single axis of rotation measured at the points of power transmission (across the coupling) when equipment is running at normal operating conditions. Misalignment causes potentially high stresses in the couplings, shafts, bearings, and seals. Additionally, shaft alignment is influenced by temperature. Although our generator had a considerable temperature rise, we ignored those effects in comparison to the huge misalignments we were introducing.

The procedure used to introduce misalignment was to first align the motor to the pump. The two prominent methods used for precision alignment of coupled shafts are the reverse indicator method and the computer-aided laser shaft alignment systems. Reverse indicator alignment procedures were used in these tests. After alignment, we shifted the motor feet in the horizontal direction with adjusting screws, noting the distance moved with dial gauges. The screws and the dial gauges can be seen in Figure 3. Parallel or offset misalignment means to shift both the near and far feet the same distance. Angular displacement can be understood with respect to Figure 4. Manufacturers give the allowable angular misalignment in terms of δ and d as shown on Figure 4. These values allow one to calculate the distance to move the near foot, d_n, and the far foot, distance, d_f, according to:

$$\frac{\delta}{d} = \frac{d_n}{x_n} = \frac{d_f}{x_f} \tag{3}$$

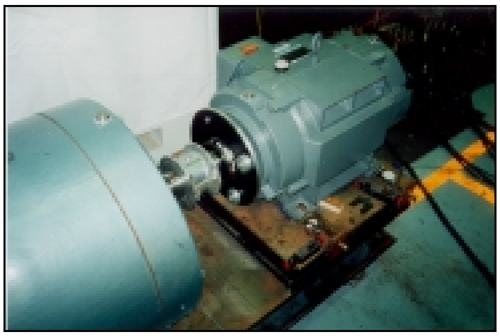

Figure 3. Motor with dial gauges and adjuster screws for alignment, balancing ring with weights for balancing, and the grid coupling in place.

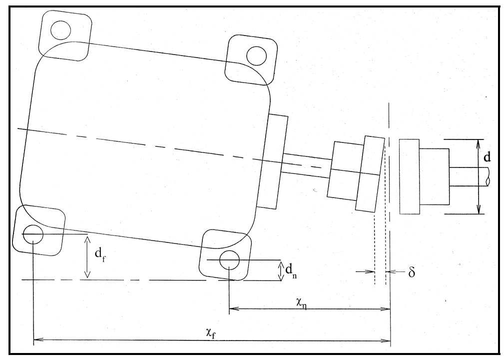

Figure 4. Angular misalignment parameters.

Coupling manufacturers indicate maximum allowable misalignment in their installation instructions. Table I gives values of the misalignment parameters from the manufacturer's literature, and gives the near and far foot moves for maximum angular misalignment calculated with Equation (3).

Table I. Test Coupling Parameters

Coupling Designation	Size	Max Offset	δ	d	d_n	d_f
Dodge Para-Flex (Boot)	PX60	0.03125	0.031	5.0	0.030	0.062
Falk Steelflex, Type T10 (Grid)	1050T	0.016	0.016	3.562	0.044	0.094
Rexnord Thomas Type DBZ-B (Shim Pack)	163	0.020	0.008	4.56	0.077	0.162
Kop-Flex Forged Steel (Gear)	2	0.004	0.002	3.25	0.006	0.013

The actual misalignments tested were greater than listed above. The concept was to test misalignments that might result from poor practice, and the object was to measure energy consumption. Basically, we found no measurable energy consumption at combinations of the above levels on both the grid and the boot couplings so we exaggerated the misalignment as much as we dared. We drilled the holes in the motor feet to ¾-inch and used ½-inch bolts with large washers to hold the motor. We clearly went too far in the case of the Shim Pack because at the end of the high power run, several of the shims had fractured. The maximum misalignment condition for the gear couplings was arrived at by consulting with a senior Kopflex (the gear coupling manufacturer) representative [7]. His recommendation was for a maximum parallel misalignment condition of 0.035-inch. The actual tested misalignments are given in Table II.

Table II. Test Misalignments

Coupling Type	Misalignment Description	d_n	d_f
Grid	Max Parallel	0.016	0.016
	Max Angular	0.044	0.094
	4 Par - 3 Ang (Test condition)	0.068	0.218
Boot	Max Parallel	0.031	0.031
	Max Angular	0.030	0.062
	4 Par - 3 Ang (Test Condition)	0.034	-0.062
Shim Pack	Max Parallel	0.020	0.020
	Max Angular	0.077	0.162
	4 Par - 2 Ang (Test Condition)	0.074	0.244
Gear	0.035 Offset (Test Condition)	0.035	0.035

Theoretical Energy Loss Explanation:
In the case of the gear and grid coupling, a corresponding colleague [8] suggested an analysis model which was adapted to sliding in a gear coupling. Figures 5a and 5b illustrate the sliding caused by misalignment that can consume energy. Figure 5a shows the sliding due to angular misalignment, and Figure 5b shows the situation for offset or parallel misalignment. Imagine that the two hubs have external gear teeth where they contact the sleeve; they are on the two shafts being coupled. The sleeve (cross hatched) has internal gear teeth and meshes with the external teeth of the hubs. Thus, one shaft transmits torque to its hub, to the sleeve, to the next hub, and to the output shaft. Note that the allowable misalignment is limited by the clearances. Both figures illustrate the maximum possible misalignment since the hubs are up against the flanges at the end of the sleeve. Notice also that the maximum angular condition in Figure 5a, allows about half the slip distance as in the offset case of Figure 5b. Since the amount of slip in parallel

or offset misalignment is twice that of angular misalignment, offset misalignment was selected as the condition to test.

In each case (offset or angular misalignment) each tooth on each hub slides over and back an amount, δ, during each revolution resulting in a total slide of, 2δ, each revolution. Work is done during this sliding and is equal to the friction force times the sliding distance, 2δ. The friction force is the normal force times the coefficient of friction, μ. The sum of all the normal forces on all of the teeth of one hub, N, is the torque, T, divided by the radius, r. The torque is given by the transmitted power, P, divided by the angular velocity , ω (radians per sec). Now writing an equation for the normal force we have:

$$N = \frac{T}{r} = \frac{P}{\omega r} \qquad (4)$$

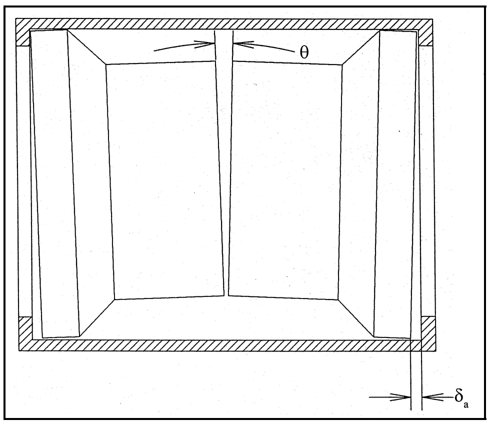

Figure 5a. Angular misalignment

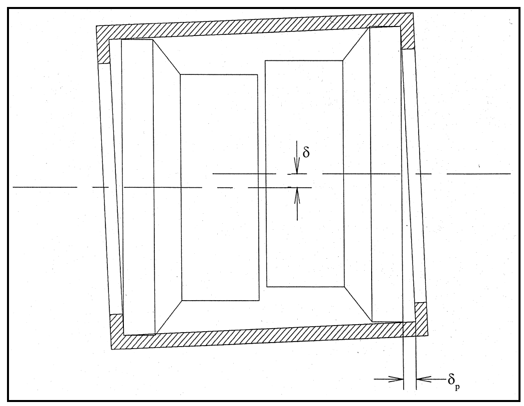

Figure 5b. Offset on parallel misalignment.

The power wasted, will be the sliding work done on each hub per revolution, ($\mu N2\delta$) , twice this for both hubs, times the revolutions per unit time or $\omega/2\pi$, or:

$$P_{wasted} = \mu N 4\delta \frac{\omega}{2\pi} = \mu \frac{P}{\omega r} 4\delta \frac{\omega}{2\pi} = \frac{\mu P 2\delta}{r\pi} \qquad (5)$$

The lost power ratio, η, is the wasted power over the transmitted power which is:

$$\eta = \frac{2\mu\delta}{\pi r} \qquad (6)$$

Substituting the typical values of $\mu = 0.1$, $\delta = 0.035$, r = 2.12-inch, we get:

$\eta = 0.00105$ or 0.1 percent loss.

This is probably high because with a lubricated coupling the coefficient of friction would be less than 0.1.

Unbalance Procedures:
General unbalance contains some fraction of both static and dynamic unbalance. For our tests we used a balance wheel where weights in the form of bolts and washers could be conveniently attached to both balance and unbalance the rotor in that single plane. The balance wheel was fabricated from a commercial sprocket with the teeth machined off. The result was a 10-inch diameter, 7/16-inch thick, disk with a hub and is shown in Figure 3. Four equally spaced threaded holes on a 4-inch radius were provided for attaching weights in the form of washers and bolts. By splitting the weight between two adjacent holes, the effective angular position of the added unbalance can be placed in any angular direction. Balancing measurements were made with a CSI Machinery Analyzer with a balancing program [9].

We planned to introduce unbalance such that the resulting unbalance force would be in increments of tenths of the rotor weight. For our motor with a 95-pound armature, at the 4-inch bolt radius of the balancing ring, this yields a weight increment of 0.413-ounce or 11.7 grams. Unbalance testing with the grid coupling on the rigid base caused very little power loss, and we added a great many of these weight increments to little avail.

Unbalance Tests:
This project has been a protracted one, largely because we expected much greater energy losses than we found. Unbalance testing was done in August of 1995. Six runs were made with unbalances ranging from 0.4-ounce, to 7 ounces; almost half a pound At that time, we were running test times of 5 to 10 minutes, and attempting to read the varying digital power outputs from the precision wattmeters, by writing as fast as possible. About 15 power measurements were recorded, efficiencies were calculated and plotted as a function of time. The values slope downward over the recording interval, due to a temperature rise in the generator which causes increased copper resistance and thus decreased efficiency. The data is inconclusive, and near the resolution of our instrumentation. The efficiencies varied between 0.780 and 0.783. This is a percentage change of efficiency of 0.003/0.783 x 100 = 0.38 percent.

That is the greatest efficiency loss one could infer from our unbalance tests, and we don't believe it is as great as that, because at that time we had not yet realized that temperature stabilized testing was important.

Power Increase And Efficiency Decrease:
Unbalance with the commercial base caused an efficiency decrease of 0.4 percent. Unbalance on the rigid base also caused a decrease of 0.4 percent.

Table III presents the results of our extreme misalignment tests on the four couplings. For the grid and the boot couplings we measured approximately a 1 percent efficiency decrease at 25 percent power. The grid coupling was severely misaligned, so much so, that it was difficult to turn the rotor by hand. After the coupling had run for 6 hours, it seemed to have worn in. Perhaps that helps explain the lower reading at 100 percent power. This certainly does not explain the zero power change for no load. The shim pack readings are all high. We over misaligned the coupling just as we were concluding that stabilized testing was needed. We could not go back and re-test the aligned condition because we fractured several of the shim discs. The boot coupling got hot, but did not seem to suffer at all, The lubricated gear coupling appeared to consume no energy at all.

Table III. Power Increase and Efficiency Decreases Due to Gross Misalignment

Coupling Type:	Shim pack	Grid	Boot	Gear
ΔP_i for P_0=0	-1.3%	0.0%	4.9%	0%
$\Delta\eta$ for P_0=25%	0.12%,	1.19%,	1.02%	0.19%
$\Delta\eta$ for P_0=100%,	0.2%	0.13%	0.24%	-0.05%

CONCLUSIONS:

1. The test program has measured less than 1.2 percent energy losses from gross misalignment, while unbalance appears to cause losses of less than 0.5 percent. Thus, a reason for precision alignment and balancing is not energy conservation.

2. The tests were conducted on the boot, grid, shim pack disk, and gear couplings at 1800 RPM and at a maximum load of 20 kW.

3. For some couplings a slight energy savings can be expected, but the effect is quite minimal. Certainly not sufficient to drastically change maintenance practices.

4. Severe misalignment on both the grid and the boot couplings caused significant temperature rise of the couplings. This observation supports the concept of using infrared thermography to identify gross misalignment, as indicated in Reference 5.

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8. Email correspondence Allen Marshall, Aukland, New Zealand July 1995

9. Downloadable Balancing Program for the Model 2110-DSP Machinery Analyzer manufactured by Computational Systems, Inc. 835 Innovation Dr.; Knoxville, TN 37932

APPENDIX:

EQUIPMENT LIST

LOAD BANK
Simplex LB180/250 Load Bank
Resistive Load Bank
WO#
Model: ADV L.B. 180/250
KW: 187/250 @ 240/480 VAC
Max. Volt: 480 VAC 250 VDC
Mfg. By Simplex Inc.
Springfield, IL

POWER METERS
Yokogawa Model 2533 Digital Power Meter
Yokogawa Type 2505 Digital AC Meter

CURRENT TRANSFORMERS
YEW Portable Current Transformers
Model J1S C1731 Type 2241
Rated Accuracy 0.2 percent
Yokogawa Electric Works, Ltd.

THERMOCOUPLE READOUT
Solomat Multi-Purpose Instrument Package
Model MPM 500E